HIGH-SPEED laser scanning technology has numerous applications, such as raster scanning for automotive head-up displays, head-worn displays [1]–[4], and other wearable displays for personal electronic devices or mobile computing. Currently, microelectromechanical systems (MEMS)-based microlaser scanners, particularly for micromirror scanners [5], [6], which utilize out-of-plane deflection to cause the laser beam to scan, were mostly developed due to their outstanding advantages compared with macroscaler scanners, such as having a low mass, a high scanning frequency, low power consumption, and a potentially low unit cost. However, due to the nature of the microfabrication process, the mirror plate is usually very thin. This brings about significant aberration to the optical system during high-speed scanning because of the dynamic nonrigid body deformation of the mirror plate under out-of-plane acceleration forces. Instead of using out-of-plane deflection, MEMS-based in-plane vibratory grating scanners [9]–[11] utilize in-plane rotation of a diffraction grating to cause the laser beam to scan. Since the nonrigid body deformation of a thin plate under in-plane excitation is much smaller than that under out-of-plane excitation, MEMS in-plane vibratory grating scanner has the potential to scan at high frequencies with little optical degradation.

Fig. 1 shows the operation principle of the vibratory grating scanner. The diffraction grating lies in the $XOY$ plane and the grating lines oriented parallel to the $X$-axis. The diffraction grating is illuminated by an incident laser beam, which lies in the $YOZ$ plane, with an incident angle of $\theta_i$. When the diffraction grating rotates about the $Z$-axis in the $XOY$ plane, the diffraction beam (except the zeroth-order beam) will scan accordingly. Bow-free scanning trajectory can be achieved when the incident angle, grating period, diffraction orders, and wavelength of incident laser beam obeys the bow-free scanning conditions [10]. High-diffraction efficiency of more than 75% can be achieved when a transverse magnetic (TM)-polarized laser beam is utilized [11].

Since diffraction grating is a dispersive optical element, a grating scanner with a single grating is only suitable for narrowband laser scanning applications, e.g., monochromatic laser scanning displays. However, by configuring multiple diffraction grating elements on a common platform, a vibratory grating scanner can also be used in multibandwidth collinear scanning applications [10], such as color displays. In addition, the optical efficiency of each wavelength can be optimized by optimizing the corresponding grating profile, which can be realized by using multistep lithography and an etching process.

In our previous work, high-speed high-optical-efficiency bow-free laser scanning without dynamic deformation has been...
successfully demonstrated with the prototype devices fabricated using silicon-on-insulator (SOI) micromachining technology [11]. The optical scan angle of the previous prototype device is limited by the maximum allowable deformation of the flexural beams due to their excessive internal stress. To further enhance the optical scanning angle, a new structure design with lower maximum internal stress in the flexural beams is required.

In this paper, we demonstrate the design, modeling, fabrication, and experimental results of an improved MEMS vibratory grating scanner driven by a novel two-degree-of-freedom (2-DOF) electrical comb-driven circular resonator, which has a much better optical performance. The prototype device with a 1-mm-diameter diffraction grating can achieve an optical scan angle of around 25° for a 632.8-nm-wavelength laser beam at its resonant frequency of 20.289 kHz.

II. 2-DOF ELECTRICAL COMB-DRIVEN CIRCULAR RESONATOR

Microcomponents in a MEMS device can be actuated by microactuators either directly or indirectly [7], [8]. High-speed in-plane rotational vibration of a micromachined diffraction grating can be realized by a one-degree-of-freedom (1-DOF) electrostatic comb-driven circular resonator, which is illustrated in Fig. 2(a). However, under this direct actuation scheme, the in-plane rotational range of the grating platform is determined by the travel range of the comb-driven circular actuator, which is limited due to the pull-in of comb fingers. This design will inevitably lead to a small scanning angle when the diameter of the grating platform is large. However, in many optical scanning applications, a larger beam size is always preferred to achieve higher optical resolution.
Fig. 3. FE simulation results of the maximum internal stress versus the number of main flexural beams when the grating platform’s rotational angle is 8°.

To overcome this problem, an improved driving mechanism using an indirect actuation method that utilizes a 2-DOF electrical comb-driven lateral-to-rotational resonator [shown in Fig. 2(b) and (c)] was reported in our previous work [9]–[11]. Under this indirect actuation scheme, the rotational motion of the grating platform is excited by the symmetric linear motions of several electrostatic comb-driven microactuators through several flexural beams. The rotational range of the grating platform is no longer determined by the travel range of microactuators but only by the mode shape design and maximum allowable deformation of the flexural suspension beams, which is the deformation when the internal maximum stress reaches their rupture stress. Without changing the total stiffness, hence, maintaining the same resonant frequency, the internal maximum stress can be reduced by reducing the width of each flexural beam and increasing the total number of suspension beams, which has been proved by finite element (FE) simulations [type of analysis: static stress analysis; element type: C3D8R; number of elements: 19 128; degrees of freedom (DOFs): 6] using the commercial software package ABAQUS. The simulation results are shown in Fig. 3. However, due to limited space, the number of flexural suspensions is typically limited, even if extra supporting springs are added [shown in Fig. 2(c)]. Consequently, the rotational range of the grating platform still cannot be very large. In addition, due to uncertainties and imperfections of the microfabrication process, the resonant frequency of each microresonator is typically different, which can cause variation of the diffraction grating’s rotational center and marked performance deviations from the original design. Although this problem can be solved by adding feedback control to each individual microresonator, the complexity of the system will increase significantly.

To further increase the rotational range of the grating platform, a novel driving mechanism that utilizes a 2-DOF electrical comb-driven circular-resonator driven through several flexural beams. The rotational range of the grating platform is no longer determined by the travel range of microactuators but only by the mode shape design and maximum allowable deformation of the flexural suspension beams, which is the deformation when the internal maximum stress reaches their rupture stress. Without changing the total stiffness, hence, maintaining the same resonant frequency, the internal maximum stress can be reduced by reducing the width of each flexural beam and increasing the total number of suspension beams, which has been proved by finite element (FE) simulations [type of analysis: static stress analysis; element type: C3D8R; number of elements: 19 128; degrees of freedom (DOFs): 6] using the commercial software package ABAQUS. The simulation results are shown in Fig. 3. However, due to limited space, the number of flexural suspensions is typically limited, even if extra supporting springs are added [shown in Fig. 2(c)]. Consequently, the rotational range of the grating platform still cannot be very large. In addition, due to uncertainties and imperfections of the microfabrication process, the resonant frequency of each microresonator is typically different, which can cause variation of the diffraction grating’s rotational center and marked performance deviations from the original design. Although this problem can be solved by adding feedback control to each individual microresonator, the complexity of the system will increase significantly.

III. SCANNER DESIGN AND MODELING

The schematic of the micromachined 2-DOF electrical comb-driven circular-resonator-driven in-plane vibratory grating scanner is illustrated in Fig. 4, where we can see that the round platform with the diffraction grating is connected to the outer comb-driven circular resonator through 16 single-beam flexures, and each of them has two pairs of perpendicularly connected stress alleviation beams, which are used to reduce its axial stress during large deformation [11], [12]. Among the 16 single-beam suspensions, eight of them are designed to be longer than the others to save space. The outer comb-driven circular resonator is suspended by symmetrically configured circular folded beam suspensions and is driven by electrostatic comb-driven circular actuators.

A. Modeling of the Main Flexural Beams

We take advantage of the stiffness matrix [13] of a single beam to obtain the model of the main flexural beams. The model of a main flexural beam and its corresponding local variables are shown in Fig. 5. In this model, when an external torque $\tau_0$ is applied, the grating platform will rotate $\theta$. The spring constant of one main flexural beam can be expressed as the ratio of $\tau_0$ to $\theta$.

The beam is assumed to have small deformations so that the axial deformation is ignored and the stress alleviation springs are not considered at the moment. The relationship between the
local variables of force \((F_1, \tau_1, F_2, \text{and } \tau_2)\) and displacement \((\nu_1, \alpha_1, \nu_2, \text{and } \alpha_2)\) for a single beam can be expressed as

\[
\begin{bmatrix}
F_1 \\
F_2 \\
\tau_2
\end{bmatrix} = \frac{EI}{L^3}
\begin{bmatrix}
12 & 6L & -12 & 6L \\
6L & 4L^2 & -6L & 2L^2 \\
-12 & -6L & 12 & -6L \\
6L & 2L^2 & -6L & 4L^2
\end{bmatrix}
\begin{bmatrix}
\nu_1 \\
\alpha_1 \\
\nu_2 \\
\alpha_2
\end{bmatrix}
\]  
(1)

where \(L\) is the beam length, \(I\) is the area moment of inertia of the beam, and \(E\) is the Young’s modulus of the material.

By applying the boundary condition shown in Fig. 5 into (1), we can obtain

\[
\begin{bmatrix}
F_1 \\
F_2 \\
\tau_2
\end{bmatrix} = \frac{EI}{L^3}
\begin{bmatrix}
12 & 6L & -12 & 6L \\
6L & 4L^2 & -6L & 2L^2 \\
-12 & -6L & 12 & -6L \\
6L & 2L^2 & -6L & 4L^2
\end{bmatrix}
\begin{bmatrix}
0 \\
0 \\
-\theta_0 \\
\theta
\end{bmatrix}
\]  
(2)

where \(\theta_0\) is the radius of the grating platform.

Because the grating platform is in rotational equilibrium (shown in Fig. 5), the external torque \(\tau_0\) can be expressed as

\[
\tau_0 = \tau_2 - F_2 R_0.
\]

From (2) and (3), we can obtain

\[
\tau_0 = \frac{EI}{L^3} (12R_0^2 + 12R_0L + 4L^2) \theta.
\]  
(4)

The spring constant for one main flexural beam is

\[
k_c = \frac{EI}{L^3} (12R_0^2 + 12R_0L + 4L^2).
\]  
(5)

Since two types of the main flexural beams are adopted, the spring constant for each type, which is expressed as \(k_{ci}\), is shown in the following:

\[
k_{ci} = \frac{EI}{L_i^3} (12R_0^2 + 12R_0L_i + 4L_i^2), \quad i = 1, 2.
\]  
(6)

Therefore, the total spring constant of the main flexural beams, which is expressed as \(K_c\), is given as follows:

\[
K_c = n_1 k_{c1} + n_2 k_{c2}
\]  
(7)

where \(n_1\) and \(n_2\) are the numbers of each type of the main flexural beams.

\[\text{Fig. 5. Model of the main flexural beam and its corresponding local variables.}\]

\[\text{Fig. 6. Schematics of the lateral folded beam suspension and the circular folded beam suspension.}\]

\[\text{Fig. 7. Model of one set of circular folded beam suspension.}\]

**B. Modeling of the Suspensions for a Circular Resonator**

In this design, we use the circular folded beams as the suspensions of the outer circular resonator. Fig. 6 shows the schematics of the lateral folded beam suspension and the circular folded beam suspension. In Fig. 6, we can see that similar to the lateral folded beam suspension, one set of circular folded beam suspension is composed of four identical single-beam flexures with one end connected to each other through a rigid truss structure. Two of them are connected to the movable structure, and the other two are connected to the fixed boundary. However, the axial lines of the four beams are no longer parallel to each other but are coincident with the rotation center, and the rigid truss structure is also changed from a rectangular shape to a sector-annular shape.

Fig. 7 shows the model of one set of circular folded beam suspension. In Fig. 7, we can see that beams 1, 2, 3, and 4 are connected in series, respectively, and the two sets of the serially connected beams are then connected in parallel. Since the dimensions of the four beams are same, the rotational spring constant of one set of circular folded beam suspension

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Fig. 8. Schematics of the simplified 2-DOF undamped vibration systems.

is equivalent to that of one single beam. According to (5), we can obtain

$$k_f = \frac{EI_f}{L_f^3} \left(12R_1^2 + 12R_1 L_f + 4L_f^2\right)$$  \hspace{1cm} (8)

where $L_f$ and $I_f$ are the length and area moment of inertia of the beam in circular folded beam suspension, respectively.

If the number of the circular folded beam suspension is $n_f$, the total spring constant of the suspensions of the circular resonator, which is expressed as $K_f$, is given as follows:

$$K_f = n_f \cdot k_f.$$  \hspace{1cm} (9)

C. Simplified Model of the Scanner

As shown in Fig. 8, the scanner can be simplified to a 2-DOF spring-mass vibration system. All the connection suspensions are considered as ideal springs whose weights are ignored.

The total kinetic and potential energy of the system during vibration, which are expressed as $E_k$ and $E_p$, respectively, are shown in the following:

$$E_k = \frac{1}{2} J_0 \dot{\theta_0}^2 + \frac{1}{2} J_1 \dot{\theta_1}^2$$  \hspace{1cm} (10)

$$E_p = \frac{1}{2} K_f \theta_1^2 + \frac{1}{2} K_c (\theta_0 - \theta_1)^2$$  \hspace{1cm} (11)

where $J_0$ and $J_1$ are the moment of inertias of the grating platform and outer circular resonator, respectively, and $R_1$ is the radius of the connection part where the circular folded beam suspension connected to.

Then, the linear model of the system is obtained by applying the Euler–Lagrange formulation [13]

$$\begin{bmatrix} J_0 & 0 \\ 0 & J_1 \end{bmatrix} \begin{bmatrix} \ddot{\theta}_0 \\ \ddot{\theta}_1 \end{bmatrix} + \begin{bmatrix} K_c & -K_c \\ -K_c & K_f + K_c \end{bmatrix} \begin{bmatrix} \theta_0 \\ \theta_1 \end{bmatrix} = 0.$$  \hspace{1cm} (12)

Through (12), we can obtain the natural frequencies and mode shapes, which can be defined as the ratio of the rotational angle of the grating platform and the outer driving circular resonator, for both the first and second vibratory modes. These can be used to get a preliminary estimation of the vibration characteristics of the system.

To ensure enough beam size of the scanned light, we attempted a design to have a 1-mm-diameter grating platform. The suspension flexural beams were designed to achieve enough scanning frequency and appropriate mode shapes. Sixteen main flexural beams were adopted, where eight beams are 15 μm wide and 1050 μm long and the rest are 18 μm wide and 1400 μm long. Two pairs of beams with a width of 7 μm and a length of 410 μm were perpendicularly connected to each main flexural beam. The outer circular resonator is suspended by 40 sets of circular folded beam suspensions with beams of width 18 μm and length 405 μm. There are 264 movable circular fingers for one side driving with finger width 7 μm, finger gap 4 μm, and initial finger overlap angle 1°. The thickness of all the structures is 80 μm. The material properties of the single crystal silicon used in this model are shown in Table I. The calculated natural frequencies and mode shapes for the proposed prototype grating scanner using the simplified dynamic model are compiled in Table II.

D. Weight Influence of the Suspension Beams

A simplified model has been obtained in Section III-B, with an assumption that the mass of all the suspension springs is ignored. However, a more rigorous model considering the effects of the mass of the suspension springs, stress alleviation beams, and fabrication imperfections is necessary to obtain a more accurate estimation. The mass of the main flexural beams and beams of the circular folded beam suspensions will be included into the dynamic model in this section.

Fig. 9 shows the model of a deformed single-beam suspension. This model essentially considers the deformation profile of the beam suspension due to the rotation of both the grating
platform and the outer circular resonator. The deflection profile
of a main flexural beam is approximated by a third-order
polynomial expression that satisfies the boundary conditions
given in the following, where all the angles are assumed to be
very small.

$$\begin{align*}
\Delta \theta_0 &= R_0 \sin \theta_0 \\
\Delta \theta_1 &= (R_0 + L) \sin \theta_1 \\
\Delta \theta_1 &= \tan \theta_0 \\
\Delta \theta_1 &= \tan \theta_1
\end{align*}$$

(13)

The function of \( \theta_0 \) and \( \theta_1 \) of the resulting deformed profile is

$$f(u) = \left( \frac{2R_0 + L}{L^3} \right) \frac{u^3}{3} - \frac{3R_0 + 2L}{L^2} \frac{u^2}{2} + u + R_0 \theta_0$$

$$+ \left( \frac{3R_0 + 2L}{L^2} \frac{u^2}{2} - \frac{2R_0 + L}{L^3} \right) \theta_1.$$

(14)

The additional kinetic energy for the main flexural beam \( \Delta E_{kc} \) can be computed by

$$\Delta E_{kc} = \int_0^L \frac{1}{2} \hat{f}^2(u) \, dm = \frac{1}{2} \rho Si WT \int_0^L \hat{f}^2(u) \, du.$$

(15)

The resulting additional kinetic energy for two types of main
flexural beams, which is expressed as \( \Delta E_{kc1} \), is shown in the following:

$$\Delta E_{kc1} = \frac{1}{2} \rho Si Wf TL_i$$

$$\times \left[ \left( \frac{13}{35} R_0^2 + \frac{11}{105} R_0 L_i + \frac{1}{105} L_i^2 \right) \dot{\theta}_0^2$$

$$+ \left( \frac{13}{35} R_0^2 + \frac{67}{105} R_0 L_i + \frac{29}{105} L_i^2 \right) \dot{\theta}_1^2$$

$$+ \left( \frac{9}{70} R_0^2 + \frac{9}{70} R_0 L_i + \frac{1}{42} L_i^2 \right) \dot{\theta}_0 \dot{\theta}_1 \right],$$

$$i = 1, 2$$

(16)

where \( \rho Si \) is the density of silicon.

Using (16), we can directly obtain the additional kinetic energy for the beams in the circular folded beam suspension, which is expressed as \( \Delta E_{kf} \), by applying corresponding boundary conditions

$$\Delta E_{kf} = \frac{1}{2} \rho Si Wf TL_f \left( \frac{87}{70} R_f^2 + \frac{41}{42} R_f L_f + \frac{67}{210} L_f^2 \right) \dot{\theta}_1^2.$$

(17)

Therefore, the modified kinetic energy of the whole system, which is expressed as \( \Delta E_k \), is then

$$\Delta E_k = \Delta E_{k1} + \Delta E_{kc1} + n_2 \Delta E_{kc2} + n_f \Delta E_{kf}.$$

(18)

By applying the Lagrange formulation again, where only the kinetic component changed, we can obtain the linear model of the system to be

$$J_0 \Delta J_0 + \Delta J_c \Delta J_c + J_1 + \Delta J_1 \begin{bmatrix} \dot{\theta}_0 \\ \dot{\theta}_1 \end{bmatrix} + \begin{bmatrix} K_c & -K_c \\ -K_c & K_f + K_c \end{bmatrix} \begin{bmatrix} \theta_0 \\ \theta_1 \end{bmatrix} = 0.$$

(19)

where

$$\Delta J_0 = \rho Si T \sum_{i=1}^2 n_i W_i L_i \left( \frac{13}{35} R_i^2 + \frac{11}{105} R_i L_i + \frac{1}{105} L_i^2 \right)$$

$$\Delta J_c = \rho Si T \sum_{i=1}^2 n_i W_i L_i \left( \frac{9}{70} R_i^2 + \frac{1}{42} R_i L_i + \frac{1}{210} L_i^2 \right)$$

$$\Delta J_1 = \rho Si T \sum_{i=1}^2 n_i W_i L_i \left( \frac{87}{70} R_i^2 + \frac{41}{42} R_i L_i + \frac{67}{210} L_i^2 \right).$$

If we ignore the mass of the main flexural beams, the grating
platform’s moment of inertia estimation error can be calculated by \( \varepsilon = \Delta J_0/(J_0 + \Delta J_0) \times 100\% \), and the calculation results show that when the number of the main flexural beams increase
from 4 to 40, the moment of inertia estimation error increases from 18.4% to 51.1%, which will introduce a significant error to the natural frequency and mode shape analysis.

FE simulations (type of analysis: natural frequency extraction; element type: C3D8R; number of elements: 109,296; DOFs: 6) investigating the natural frequencies and mode shapes of the system were conducted to verify whether the approximations were reasonable using ABAQUS. Fig. 10 shows the FE simulation results. The results obtained from the simplified model [see (12)], modified model considering the mass of the suspensions [see (19)], and FE simulation were compared in Table II.

As shown in Table II, the model considering the mass of the beams shows much better approximation of natural frequencies and mode shapes. Generally, the mass of the main flexural beams and beams in circular folded beam suspensions will reduce the effective frequencies in the device as expected. In addition to a significant increase in the mode shape for the first resonating mode and a decrease in the second mode, there is a significant frequency drop for the first resonating mode compared to the second mode. The reason is that the moment of inertia of the grating platform is much smaller than the outer circular resonator, ignoring the mass of the suspension beams has a greater influence on the frequency of the grating platform.

### E. Influence of the Stress Alleviation Beams

The stress alleviation beams were transversely connected to the main flexural beam at the junction point to help reduce the nonlinearity of the device during operation. Ideally, only translational motion of the junction point along the axial direction of the main flexural beam is allowed to release its axial stress, and the rotation of the junction point is not allowed so that the bottom boundary condition of the main flexural
beam will not change. Therefore, adding a set of ideal stress alleviation beams has little influence on the linear harmonic analysis. However, the actual stress release springs cannot fully avoid the rotational motion of the junction point, hence, the rotational spring constant was reduced due to the small tilting angle at the anchor location, as shown in Fig. 11(a).

FE simulations (type of analysis: static stress analysis; element type: C3D8R; number of elements: 19 128; DOFs: 6) considering the geometric nonlinearity using ABAQUS were done to show that the linearity of the torque–angle relationship is greatly improved when the stress alleviation beams were added [see Fig. 12(b) and (c)]. More importantly, the linear analysis of the torque–angle relationships with and without the stress alleviation beams shows the reduction of the rotational spring constant that we have to account for.

Define $\delta_1$ and $\delta_2$ as the reduction ratio of the rotational spring constant of two types of the main flexural beams, respectively, which can be directly obtained from the FE simulation results. We can then obtain the refined model considering the spring constant reduction as

$$
\left[ \begin{array}{cc}
J_0 + \Delta J_0 & \Delta J_e \\
\Delta J_e & J_1 + \Delta J_1 
\end{array} \right] \left[ \begin{array}{c}
\tilde{\theta}_0 \\
\tilde{\theta}_1 
\end{array} \right] + \left[ \begin{array}{cc}
K_c & -\tilde{K}_c \\
-\tilde{K}_c & K_f + \tilde{K}_c 
\end{array} \right] \left[ \begin{array}{c}
\theta_0 \\
\theta_1 
\end{array} \right] = 0
$$

(20)

where $\tilde{K}_c = n_1k_{c1}\delta_1 + n_2k_{c2}\delta_2$.

FE simulations (type of analysis: natural frequency extraction; element type: C3D8R; number of elements: 182 844; DOFs: 6) investigating the natural frequencies and mode shapes of the system were again done to verify the overall effects on the system using ABAQUS. Fig. 12 shows the simulation results. The predicted natural frequencies and mode shapes obtained from the model considering the mass of the beams [see (19)], the model considering the mass of the beams as well as the influence of stress alleviation beams [see (20)], and FE simulation were compared in Table III.

In Table III, we can see the similar effect of the stress alleviation beams and the mass of the beams in all the suspensions in Section III-D: both change the mode shapes and reduce the natural frequencies. However, the effective spring constant of the vibration structures was reduced instead of increasing its moment of inertia in Section III-D. The effects for the mode shapes are equally significant for both the first and second resonating modes. In Table III, we can see that the theoretical model considering both the mass of the beams and the effect of the stress alleviation beams shows closer agreement to the FE simulation results.

### F. Influence of Fabrication Imperfections

Fabrication imperfection is another very important issue that we need to address in the rigorous model. The imperfections in the plasma etching process (such as etching slope, undercut, and notching effect) will change the dimensions and area moment of the beams. This can induce a significant change in the stiffness of the suspension beams and finally bring a significant error to the predictions of the natural frequency and mode shape. Fig. 13 shows a cross-sectional profile model of the beam after a deep reactive ion etching (DRIE) process and its corresponding SEM image obtained from the previous process.

The etching undercut $\Delta W_1$ and the sidewall etching slope $\alpha$ have been included in this cross-sectional model. In addition,
this cross-sectional model attempts to account for the notching effect, which is expressed by the thickness $T_n$ and sidewall slope $\alpha_n$ of the notching area. Since the trenches of every suspension beams are the same, the parameters in the model can be directly obtained or calculated by using the measured width of the trench at different positions. As shown in the SEM image in Fig. 13, we measure the width of the trench at its top, bottom, and upper boundaries as well as the thickness of the notching area. Therefore, we can obtain

$$\tilde{W}_i = W - 2\Delta \tilde{W}_i, \quad i = 1, 2, 3 \quad (21)$$

where $\tilde{W}_i$ is the beam width at different positions shown in Fig. 13.

The etching slope and sidewall slope of the notching area can be calculated as

$$\begin{align*}
\alpha &= \arctg \left( \frac{T - T_n}{\Delta W_2 - \Delta W_1} \right) \\
\alpha_n &= \arctg \left( \frac{T_n}{\Delta W_5 - \Delta W_2} \right).
\end{align*} \quad (22)$$

Then, the actual beam width $W(u)$ can be expressed as a function of beam thickness, i.e.,

$$\begin{align*}
W(u) = \tilde{W}_3 + 2u \cdot \text{ctg} \alpha_n, \quad 0 \leq u < T_n \\
W(u) = \tilde{W}_2 + 2u \cdot \text{ctg} \alpha, \quad T_n \leq u < T.
\end{align*} \quad (23)$$

Therefore, the new area moment of inertia based on the above model, which is expressed as $\tilde{I}$, is then

$$\tilde{I} = \frac{1}{96} t g \alpha_n \left( \tilde{W}_4^2 - \tilde{W}_3^2 \right) + \frac{1}{96} t g \alpha \left( \tilde{W}_4^2 - \tilde{W}_2^2 \right). \quad (24)$$

The calculated natural frequencies and mode shapes using all the models that we had explored so far are given in Table IV. We can see the decline of the natural frequencies when more detailed compensation is added into models.

Compared to the initial simplified model, consideration of the mass of the suspension beams, stiffness reduction of the stress alleviation beams, and fabrication imperfections all reduce the theoretical predictions of natural frequencies. Significant changes also occur while predicting the mode shapes. Since the width difference between the main flexural beams and beams in circular folded beam suspensions is not significant, the effects of the stiffness reduction of the suspensions due to fabrication imperfections to the natural frequencies and mode shapes of two resonating modes are almost equal. Comparing all the models, the overall effect of the mass of the suspension beams is the most significant. This is because the multiple beam suspensions were adopted, which is used to reduce the maximum stress, hence, the total rotational moment of inertia of the beams is too large to be ignored. Ignoring the weight influence of the beams will introduce a large estimation error to the grating

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**Table IV**

**Summary of Theoretical Results Using Different Models**

<table>
<thead>
<tr>
<th>Mode</th>
<th>Simplified Model</th>
<th>Model considering the mass of the beams</th>
<th>Model considering the mass of the beams and the release spring</th>
<th>Model considering the mass of the beams, the release spring and fabrication imperfections</th>
</tr>
</thead>
<tbody>
<tr>
<td>1&lt;sup&gt;st&lt;/sup&gt; Mode</td>
<td>Frequency (Hz)</td>
<td>30245</td>
<td>24980</td>
<td>24435</td>
</tr>
<tr>
<td></td>
<td>Mode shape</td>
<td>15.35</td>
<td>111.23</td>
<td>126.00</td>
</tr>
<tr>
<td>2&lt;sup&gt;nd&lt;/sup&gt; Mode</td>
<td>Frequency (Hz)</td>
<td>32013</td>
<td>30950</td>
<td>30928</td>
</tr>
<tr>
<td></td>
<td>Mode shape</td>
<td>-28.63</td>
<td>-2.86</td>
<td>-2.56</td>
</tr>
</tbody>
</table>
platform’s moment of inertia and will, therefore, significantly influence the prediction results of the natural frequencies and mode shapes of both the first and second resonating modes.

IV. FABRICATION PROCESS

SOI micromachining technology was used to fabricate the prototype device, and four photo masks were used. The fabrication process flow is illustrated in Fig. 14. The SOI wafer used has an 80-μm-thick heavily doped silicon device layer, 2-μm-thick buried oxide (BOX) layer, and a 650 ± 25-μm-thick silicon substrate. The overall die size is 6.5 mm × 6.5 mm.

As shown in Fig. 14, the diffraction grating with a 400-nm grating period and a 50% duty cycle was patterned using deep-ultraviolet lithography and etched using timed plasma etching. The etching time is strictly controlled so that the depth of the grating groove is around 150 nm. Then, 1 μm undoped silicon glass was deposited and patterned by using a reactive ion etching (RIE) process. Next, the SOI wafer was patterned on the backside followed by a DRIE process, which is used to remove silicon and expose the region of all the structures. The etching process stopped at the BOX layer. Subsequently, the 80-μm-thick silicon device layer was etched by another DRIE process, which is also stopped at the BOX layer, to form the grating platform, comb-driven circular actuator, and suspension beams. After that, the structures formed in the SOI device layer were release from backside by using a buffered oxide etchant solution, with six parts of 40% NH₄F and one part of 49% hydrofluoric acid. Then, the metal pads for wire bonding were formed by evaporating 1000 Å/5000 Å thick Ti/Au layer through a shadow mask. Finally, a 100 Å/800 Å Ti/Au layer was evaporated on the wafer surface to enhance the reflectivity of the diffraction grating.

The whole view and the center part of the fabricated device are shown by a microscope image and an SEM image in Figs. 15 and 16, respectively.

V. EXPERIMENTAL RESULTS

We use a linearly TM-polarized He–Ne laser beam with a wavelength of 632.8 nm to test the optical performance of the
MEMS grating scanner, and the schematic of the experimental setup is illustrated in Fig. 17. Since the grating period of the diffracting grating that we adopted is 400 nm, the incident angle was determined to be $71.8^\circ$ so that bow-free scanning conditions [10] are fulfilled.

The dynamic performance of the MEMS vibratory grating scanner was tested in atmosphere and vacuum. As expected, two resonating modes exist. The optical scan angle was measured through measuring the length of the laser scanning trajectory on the projection screen, which was aligned perpendicularly to the first-order diffracted beam when the grating is motionless. The outer comb-driven circular resonator was driven by a push–pull mechanism [14] both in atmosphere and vacuum. While tested in atmosphere (760 torrs), the driving voltage was fixed at 80-V dc bias and 160-V ac peak-to-peak, and further increasing the driving power may cause the instability of the electrostatic comb-driven circular actuator. Fig. 18(a) and (b) shows the measured frequency responses in atmosphere, and the resonant frequencies of the first and second resonating modes were experimentally determined as 20.182 and 21.910 kHz with an optical scan angle of $20.8^\circ$ and $18.1^\circ$, respectively.

Fig. 18. Measured frequency response of the MEMS grating scanner in atmosphere at frequency regions near the resonant frequencies of (a) the first vibration mode and (b) the second vibration mode.

Fig. 19. Measured frequency response of the MEMS grating scanner in vacuum at frequency regions near the resonant frequencies of (a) the first vibration mode and (b) the second vibration mode.

Fig. 20. Photograph of the experimental setup for the prototype scanner.
While tested in vacuum (0.12 mtohrs), the driving voltage was fixed at 15 V dc bias and 30 V ac peak-to-peak, and further increasing the driving power may cause the brittle fracture of the main flexural suspension beams. Fig. 19(a) and (b) shows the measured frequency responses in vacuum and the resonant frequencies of the first and second resonating modes were experimentally determined as 20.289 and 21.918 kHz with an optical scan angle of 24.8° and 18.2°, respectively.

According to the dynamic testing results, operating in vacuum and scanning at the frequency near the resonant frequency of the first resonating mode are preferred due to lower driving voltage, higher scanning amplitude, and less risk of brittle fracture of the main flexural suspension beams. High-speed laser scanning was experimentally demonstrated, and Fig. 20 shows a photograph of the projected laser scanning trajectory on a projection screen, which is located at a distance of 100 mm from the grating scanner.

The scanner demonstrated some level of large-deflection nonlinearity during the vibration, e.g., the scanner’s optical scanning angle is different during the forward and backward frequency sweeping. Additionally, there are slight differences between measured resonant frequencies in atmosphere and vacuum. The linearity of the vibration can be further improved by reducing the stiffness of the stress alleviation beams along the axial direction.

In addition, the resonant frequencies and mode shapes of the prototype device were also measured on probe station under a microscope in atmosphere. The driving power was selected to be 50 V dc bias and 100 V ac peak-to-peak voltages to avoid the axial direction.

As shown in Table V, we can see that the modified rigorous model considering the mass of the suspension beams, stress alleviation beams, and fabrication imperfections can give a more accurate prediction of both resonant frequencies and mode shapes. The deviations between the theoretical and experimental results are mainly due to the uncertainty in the fabrication process, material properties, and imperfect boundary conditions of all the suspension, which is acceptable. Additionally, the measured resonant frequencies vary with different driving conditions, which is mainly due to the large-deflection nonlinearity during the vibration and different viscous damping in atmosphere.

VI. CONCLUSION

A prototype high-speed laser scanner using a vibratory grating driven by a 2-DOF electrical comb-driven circular resonator has been successfully demonstrated. When illuminated by a polarized He–Ne laser beam (632.8 nm) with an incident angle of 71.8°, the current prototype grating scanner is capable of scanning at a frequency of 20.289 kHz with an optical scan angle of around 25°. Some level of vibration nonlinearity appears during the operation due to the large deflection of the main flexural beams, which can be improved by reducing the stiffness of the stress alleviation beams along the axial direction.

We have also demonstrated the validity of a rigorous dynamic model that considers the effects of the mass of the suspension beams, stress alleviation beams, and fabrication imperfections. All affects the system significantly. However, for the current system configuration, since the multiple beam suspensions is adopted, the influence of the mass of the suspension beams was shown to have the largest effect on the system’s characteristics. Nevertheless, all of the issues were shown to affect to certain degrees, and for the future designs, we would have to take all of them into account. This paper provides a more accurate platform on which future designs can be based.

REFERENCES

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